[Name of Document] CLAIM FOR PATENT

[Claim 1]

A twin-clutch manual transmission comprising a first input shaft and a second input shaft selectively input with rotation of an engine through individual clutches, wherein the second input shaft is made hollow and rotatably fitted over the first input shaft,

the first input shaft is projected from a rear end, remote from the engine, of the second input shaft, gear pairs of one of classified drive range groups are provided so as to be properly transmittable, respectively, between a projected rear-end portion of the first input shaft and a countershaft disposed parallel to the first and second input shafts,

gear pairs of the classified other drive range group are provided so as to be properly transmittable, respectively, between the second input shaft and the countershaft, and

the rotation after a shift corresponding to a selected drive range is output in an axial direction from a rear end of the first input shaft or the countershaft,

said twin-clutch manual transmission characterized in that the gear pairs of said other drive range group are provided between said second input shaft and said countershaft such that the gear pair for a lowest-speed drive range in drive ranges in each of which a gear on said second input shaft has an outer diameter capable of

providing a predetermined bearing reception space between said first input shaft and said second input shaft is disposed on a side farthest from the engine, while the gear pair for a highest-speed drive range in the other drive ranges is disposed on a side closest to the engine.

[Claim 2]

A twin-clutch manual transmission according to claim 1,

characterized in that the gear pairs for the other drive ranges disposed between the gear pair for the drive range disposed on the side farthest from the engine and the gear pair for the drive range disposed on the side closest to the engine are such that the gear pair for the higher-speed drive range is disposed closer to the engine.

[Claim 3]

A twin-clutch manual transmission according to claim 1 or 2, wherein said other drive range group provided between said second input shaft and said countershaft is a group of even-number drive ranges,

characterized in that the gear pair of a fourth drive is disposed on the side farthest from the engine.

[Claim 4]

A twin-clutch manual transmission according to any one of claims 1 to 3,

characterized in that mesh mechanisms for causing the gear pairs of said other drive range group provided between said second input shaft and said countershaft to be

properly transmittable, respectively, are all disposed on a side of said countershaft.

[Claim 5]

A twin-clutch manual transmission according to any one of claims 1 to 4,

characterized in that a mesh mechanism dedicated for causing the gear pair disposed on the side closest to the engine to be properly transmittable is disposed between the gear pair disposed on the side closest to the engine and the gear pair disposed adjacent thereto.

[Claim 6]

A twin-clutch manual transmission according to claim 4 or 5, wherein said other drive range group provided between said second input shaft and said countershaft is a group of a second drive, a fourth drive, and a sixth drive,

characterized in that the gear pair of the fourth drive is disposed on the side farthest from the engine, the gear pair of the sixth drive is disposed on the side closest to the engine, the gear pair of the second drive is disposed between these gear pairs, and

as said mesh mechanisms disposed on the side of said countershaft, the mesh mechanism for both the second drive and the fourth drive is disposed between the gear pair of the second drive and the gear pair of the fourth drive, and the mesh mechanism dedicated for the sixth drive is disposed between the gear pair of the second drive and the gear pair of the sixth drive.

[Name of Document] SPECIFICATION

[Title of Invention] TWIN-CLUTCH MANUAL TRANSMISSION

[Technical Field]

[0001]

The present invention relates to a so-called twin-clutch manual transmission that is provided with automatic clutches for grouped drive ranges between an engine and the manual transmission and is useful for effecting automatic shifting by switching between engagement and disengagement (switching control) of those automatic clutches and alternate selection of the drive ranges between both drive range groups.

[Background Art]

[0002]

As such a twin-clutch manual transmission, there has conventionally been known a twin-clutch manual transmission for a front engine, front-wheel drive vehicle (FF vehicle) as described, for example, in Patent Document 1, wherein there are provided a first input shaft and a second input shaft selectively input with rotation of an engine through individual automatic clutches, the second input shaft is made hollow and rotatably fitted over the first input shaft, the first input shaft is projected from a rear end, remote from the engine, of the second input shaft, gear pairs of one of classified drive range groups are provided so as to be properly transmittable, respectively, between a projected rear-end portion of the first input shaft and a

countershaft disposed parallel to the first and second input shafts, gear pairs of the classified other drive range group are provided so as to be properly transmittable, respectively, between the second input shaft and the countershaft, and the rotation after a shift corresponding to a selected drive range is output in a radial direction from a front end, near the engine, of the countershaft.

[0003]

In the case of such a twin-clutch manual transmission, it is possible to effect automatic shifting by a so-called switching control of the clutches wherein, in the state where a drive range in one drive range group is selected and the corresponding automatic clutch is engaged, no drive range in the other drive range group is selected, while, when shifting, in the state where a drive range in the other drive range group is selected and the corresponding automatic clutch is disengaged, the automatic clutch associated with the foregoing one drive range group is released while the automatic clutch associated with the foregoing other drive range group is engaged, and by alternate selection of the drive ranges between both drive range groups. Therefore, although it is the manual transmission, the automatic shifting thereof is enabled.

[Patent Document 1]

Unexamined Patent Publication No. Hei 8-320054

[Disclosure of the Invention]

[Problem to be Solved by the Invention]

[0004]

On the other hand, since the FF-vehicle twin-clutch manual transmission takes out the rotation after shifting in the radial direction from the front end of the countershaft near the engine as described above, and further in view of convenience of assembly of the transmission, it is preferable that the diameter of the countershaft be configured to be the maximum at its front end and gradually decrease toward its rear end as illustrated in Patent Document 1.

[0005]

However, in a twin-clutch manual transmission for a front engine, rear-wheel drive vehicle (FR vehicle), since rotation after a shift corresponding to a selected drive range is required to be output in an axial direction from a rear end, remote from an engine, of a first input shaft or a countershaft, and further in view of convenience of assembly of the transmission and a strength of the countershaft, it is preferable that the diameter of the countershaft be configured to be the maximum at its intermediate portion (near a rear end of a second input shaft) and gradually decrease from this intermediate portion toward its front and rear ends.

[0006]

Therefore, there is a problem in using the countershaft of the conventional FF-vehicle twin-clutch manual transmission described in Patent Document 1 as the

countershaft of the FR-vehicle twin-clutch manual transmission.

Incidentally, at the time of the filing of this application, the applicant of the application is not aware of any information that the FR-vehicle twin-clutch manual transmission has conventionally been proposed.

[0007]

On the other hand, only for satisfying the requirement that the diameter of the countershaft be configured to be the maximum at its intermediate portion and gradually decrease from this intermediate portion toward its front end, since gear pairs of a drive range group disposed between the second input shaft and the countershaft are configured such that the outer diameter of a gear on the countershaft becomes smaller for a higher-speed drive range, it is sufficient that the gear pairs for the higher-speed drive ranges be arranged closer to the front side near the engine.

That is, since the small-diameter gear for the high-speed drive range is provided at the front end of the countershaft near the engine and the outer diameters of the gears on the countershaft increase as going away from the engine, it is possible to satisfy the requirement that the diameter of the countershaft be configured to be the maximum at its intermediate portion and gradually decrease from this intermediate portion toward its front end.

[0008]

On the other hand, when supporting the first input shaft and the second input shaft rotatably fitted thereover on a transmission case, the front end of the second input shaft near the engine is supported by the transmission case through a bearing and the rear end of the first input shaft projected from the second input shaft is supported by the transmission case through a bearing, thereby supporting the first input shaft and the second input shaft in the transmission case, as also shown in Patent Document 1.

Note that when, as described above, the gear pair for the lowest-speed drive range is disposed on the side farthest from the engine, since the outer diameter of a gear on the second input shaft forming that gear pair is the smallest, this gear restricts the size of an annular space defined between the rear end of the second input shaft and the first input shaft.

[0009]

In this case, a bearing such as a needle bearing cannot be received in the annular space between the rear end of the second input shaft and the first input shaft.

However, if an annular groove for receiving therein the needle bearing is formed on the first input shaft, a reduction in strength of the first input shaft is resulted and therefore this measure is difficult to take.

Instead of it, it may be considered to move the needle bearing toward the engine side up to a position where no gears are present. In this case, however, the

needle bearing approaches a bearing disposed between the front end of the second input shaft and the front end of the first input shaft to thereby reduce a bearing span so that the bearing rigidity between both input shafts becomes insufficient, and therefore, this measure is also difficult to take.

[0010]

Under the circumstances as described above, it is an object of the present invention to provide a twin-clutch manual transmission, particularly useful for a FR vehicle, which can receive a bearing in an annular space between a rear end of a second input shaft and a first input shaft without forming a bearing reception annular groove on the first input shaft by properly disposing in an axial direction gear pairs of a drive range group provided between the second input shaft and a countershaft, and which can satisfy the foregoing requirement that the diameter of the countershaft be configured to be the maximum at its intermediate portion and decrease toward its front end.

[Means for Solving the Problem]

[0011]

For this object, a twin-clutch manual transmission according to the present invention is as follows, as recited in claim 1.

First, the twin-clutch manual transmission as a basis comprises a first input shaft and a second input shaft

selectively input with rotation of an engine through individual clutches, wherein the second input shaft is made hollow and rotatably fitted over the first input shaft,

the first input shaft is projected from a rear end, remote from the engine, of the second input shaft, gear pairs of one of classified drive range groups are provided so as to be properly transmittable, respectively, between a projected rear-end portion of the first input shaft and a countershaft disposed parallel to the first and second input shafts,

gear pairs of the classified other drive range group are provided so as to be properly transmittable, respectively, between the second input shaft and the countershaft, and

the rotation after a shift corresponding to a selected drive range is output in an axial direction from a rear end of the first input shaft or the countershaft.

[0012]

With respect to such a twin-clutch manual transmission, the gear pairs of the foregoing other drive range group provided between the second input shaft and the countershaft are disposed in the following manner in the present invention.

Specifically, the gear pair for a lowest-speed drive range in drive ranges in each of which a gear on the second input shaft has an outer diameter capable of providing a predetermined bearing reception space between the first

input shaft and the second input shaft is disposed on a side farthest from the engine,

while the gear pair for a highest-speed drive range in the other drive ranges is disposed on a side closest to the engine.

[Effect of the Invention]

[0013]

According to such a twin-clutch manual transmission of the present invention, in the arrangement of the gear pairs of the drive range group provided between the second input shaft and the countershaft,

the gear pair for the lowest-speed drive range in the drive ranges in each of which the gear on the second input shaft has the outer diameter capable of providing the predetermined bearing reception space between the first input shaft and the second input shaft is disposed on the side farthest from the engine,

so that it is possible to ensure the predetermined bearing reception space between the rear end of the second input shaft and the first input shaft without forming a bearing reception annular groove on the first input shaft and therefore it is possible to receive a bearing between the rear end of the second input shaft and the first input shaft without lowering the strength of the first input shaft.

Consequently, according to the present invention, the bearing span of bearings disposed between both input shafts

can be set large to thereby ensure high bearing rigidity between both input shafts.

[0014]

Further, according to the present invention, in the arrangement of the gear pairs of the drive range group provided between the second input shaft and the countershaft, the gear pair for the highest-speed drive range in the drive ranges other than the above is disposed on the side closest to the engine,

so that the front end of the countershaft near the engine can be made small in diameter since the gear on the countershaft forming such a gear pair has a small diameter because of it being for the highest-speed drive range,

and therefore, it is possible to satisfy the foregoing requirement in terms of the assembly and also the strength that the diameter of the countershaft be configured to decrease from its intermediate portion toward its front end.

[Best Mode for Carrying Out the Invention]

[0015]

Hereinbelow, a mode embodying the present invention will be described in detail based on an embodiment shown in the drawings.

Fig. 1 shows a diagram of essentials of a twin-clutch manual transmission according to one embodiment of the present invention, and Fig. 2 shows a substantial structural diagram of the twin-clutch manual transmission.

In this embodiment, the twin-clutch manual transmission has the following structure which is useful for a front engine, rear-wheel drive vehicle (FR vehicle).

[0016]

In the figure, 1 denotes a transmission case and, as shown in Fig. 1, an automatic clutch C1 for odd-number drive ranges (first drive, third drive, fifth drive, reverse) and an automatic clutch C2 for even-number drive ranges (second drive, fourth drive, sixth drive) are interposed between a later-described gear shift mechanism received in the transmission case 1 and an engine (only a crankshaft 2 is shown in Fig. 1), and

both clutches C1 and C2 are connected to the engine crankshaft 2 through a torsional damper 3 in a damped manner.

An oil pump 4 constantly driven by the engine through the torsional damper 3 is further provided in the transmission case 1, and a later-described drive range selection control including an engagement control of the clutches C1 and C2 is executed using a hydraulic oil from the oil pump 4 as a medium.

[0017]

The gear shift mechanism received in the transmission case 1 will be described hereinbelow with reference also to Fig. 2. The gear shift mechanism comprises a first input shaft 5 and a second input shaft 6 selectively input with rotation of the engine from the torsional damper 3 through

the odd-number drive range clutch C1 or the even-number drive range clutch C2.

The second input shaft 6 is hollow and is fitted over the first input shaft 5. A front-side needle bearing 7 and a rear-side needle bearing 8 are disposed in annular spaces between both shafts, respectively, to thereby allow the inside first input shaft 5 and the outside second input shaft 6 to be mutually rotatable in a concentric state.

[0018]

Front ends, on the engine side, of the first input shaft 5 and the second input shaft 6 fitted together to be mutually rotatable as described above pass through a front wall 1a of the transmission case 1 so as to be coupled to the corresponding clutches C1 and C2, respectively.

The outer periphery of the front end of the second input shaft 6 is rotatably supported by the front wall 1a of the transmission case 1 through a ball bearing 9 and the foregoing front-side needle bearing 7 is disposed in the neighborhood thereof, while the rear-side needle bearing 8 is disposed at a rear end, remote from the engine, of the second input shaft 6.

The first input shaft 5 is projected from the rear end of the second input shaft 6 and a projected rear-end portion 5a of the first input shaft 5 passes through an intermediate wall 1b of the transmission case 1. At this passing-through portion, the rear-end portion 5a of the first input shaft 5 is rotatably supported by the

intermediate wall 1b of the transmission case 1 through a ball bearing 10.

[0019]

An output shaft 11 is provided so as to be coaxially butted to the rear-end portion 5a of the first input shaft 5. The output shaft 11 is rotatably supported by a rear-end wall 1c of the transmission case 1 through a tapered roller bearing 12 and an axial bearing 13 and is further rotatably supported on the rear-end portion 5a of the first input shaft 5 through a needle bearing 14.

A countershaft 15 is disposed parallel to the first input shaft 5, the second input shaft 6, and the output shaft 11 and rotatably supported by the front-end wall 1a, the intermediate wall 1b, and the rear-end wall 1c of the transmission case 1 through roller bearings 16, 17, and 18.

A counter gear 19 is provided at a rear end of the countershaft 15 so as to be rotatable with the countershaft 15, while the output shaft 11 is provided with an output gear 20 that is disposed in the same plane, as the counter gear 19, perpendicular to axes thereof. The counter gear 19 and the output gear 20 are meshed with each other so that the countershaft 15 is drivingly connected to the output shaft 11.

[0020]

Between the rear-end portion 5a of the first input shaft 5 and the countershaft 15, gear pairs of an odd-number drive range group (first drive, third drive,

reverse) are provided, that is, a first drive gear pair G1, a reverse gear pair GR, and a third drive gear pair G3 are disposed in the order named from the front side near the engine.

The first drive gear pair G1 and the reverse gear pair GR are disposed between the rear end of the second input shaft 6 and the transmission case intermediate wall 1b, while the third drive gear pair G3 is disposed on the opposite side of the transmission case intermediate wall 1b in close proximity thereto.

[0021]

The first drive gear pair G1 is composed of a first drive input gear 21 formed integral with the rear-end portion 5a of the first input shaft 5 and a first drive output gear 22 rotatably mounted on the countershaft 15, which gears are meshed with each other.

The reverse gear pair GR is composed of a reverse input gear 23 formed integral with the rear-end portion 5a of the first input shaft 5, a reverse output gear 24 rotatably mounted on the countershaft 15, and a reverse idler gear 25 meshed with the gears 23 and 24 to drivingly connect between the gears 23 and 24 so as to provide opposite rotation therebetween. The reverse idler gear 25 is rotatably supported on a shaft 25a secured to the transmission case intermediate wall 1b.

The third drive gear pair G3 is composed of a third drive input gear 26 rotatably mounted on the rear-end

portion 5a of the first input shaft 5 and a third drive output gear 27 drivingly connected to the countershaft 15, which gears are meshed with each other.

[0022]

The countershaft 15 is further provided with a first drive-reverse synchromesh mechanism 28 disposed between the first drive output gear 22 and the reverse output gear 24.

When a coupling sleeve 28a thereof is moved leftward from an illustrated neutral position to be meshed with a clutch gear 28b, the first drive output gear 22 is drivingly connected to the countershaft 15 so that the first drive becomes selectable as will be described later, and

when the coupling sleeve 28a is moved rightward from the illustrated neutral position to be meshed with a clutch gear 28c, the reverse output gear 24 is driving connected to the countershaft 15 so that the reverse becomes selectable as will be described later.

[0023]

The rear-end portion 5a of the first input shaft 5 is further provided with a third drive-fifth drive synchromesh mechanism 29 disposed between the third drive input gear 26 and the output gear 20.

When a coupling sleeve 29a thereof is moved leftward from an illustrated neutral position to be meshed with a clutch gear 29b, the third drive input gear 26 is drivingly connected to the first input shaft 5 so that the third

drive becomes selectable as will be described later, and

when the coupling sleeve 29a is moved rightward from the illustrated neutral position to be meshed with a clutch gear 29c, the first input shaft 5 is directly connected to the output gear 20 (output shaft 11) so that the fifth drive becomes selectable as will be described later.

[0024]

Between the hollow second input shaft 6 and the countershaft 15, gear pairs of an even-number drive range group (second drive, fourth drive, sixth drive) are provided, that is, a sixth drive gear pair G6, a second drive gear pair G2, and a fourth drive gear pair G4 are disposed in the order named from the front side near the engine.

The sixth drive gear pair G6 is disposed along the front wall 1a of the transmission case 1 at the front end of the second input shaft 6, the fourth drive gear pair G4 is disposed at the rear end of the second input shaft 6, and the second drive gear pair G2 is disposed at a center portion of the second input shaft 6 between its both ends.

[0025]

The sixth drive gear pair G6 is composed of a sixth drive input gear 30 formed integral with the outer periphery of the second input shaft 6 and a sixth drive output gear 31 rotatably mounted on the countershaft 15, which gears are meshed with each other.

The second drive gear pair G2 is composed of a second

drive input gear 32 formed integral with the outer periphery of the second input shaft 6 and a second drive output gear 33 rotatably mounted on the countershaft 15, which gears are meshed with each other.

The fourth drive gear pair G4 is composed of a fourth drive input gear 34 formed integral with the outer periphery of the second input shaft 6 and a fourth drive output gear 35 rotatably mounted on the countershaft 15, which gears are meshed with each other.

[0026]

Here, a description will be given of the reason why the gear pairs G2, G4, and G6 of the even-number drive range group (second drive, fourth drive, sixth drive) provided between the second input shaft 6 and the countershaft 15 are disposed in the manner as described above, that is, the reason for disposing the sixth drive gear pair G6, the second drive gear pair G2, and the fourth drive gear pair G4 in the order named from the front side near the engine.

[0027]

In the arrangement of the gear pairs G2, G4, and G6 of the even-number drive range group (second drive, fourth drive, sixth drive),

taking into account the foregoing requirement that the rear needle bearing 8 of the needle bearings 7 and 8 interposed between the first and second input shafts 5 and 6 is preferably located near the rear end of the second

input shaft 6 in view of the bearing span and the foregoing requirement that, in view of the strength and the assembly of the gears, the countershaft 15 preferably has the shape that the diameter thereof is the maximum at its intermediate portion corresponding to a boundary position between the even-number drive range group (second drive, fourth drive, sixth drive) and the odd-number drive range group (first drive, third drive, fifth drive, reverse) and gradually decreases toward its front end,

first, selection is made of the drive ranges (sixth drive and fourth drive) relating to the input gears 30 and 34 each having an outer diameter capable of providing a bearing reception space for the needle bearing 8 between the first input shaft 5 and the second input shaft 6, among the input gears 30, 32, and 34 formed on the second input shaft 6, and the gear pair G4 of the lowest-speed drive range (fourth drive) of the selected drive ranges is disposed on the side farthest from the engine,

the gear pair G6 of the highest-speed drive range (sixth drive) of the other drive ranges (sixth drive and second drive) is disposed on the side closest to the engine, and

the gear pair G2 of the remaining drive range (second drive) is disposed between the gear pairs G4 and G6 on both sides.

[0028]

In the illustrated example, since the number of the

even-number drive ranges is three, i.e. the second drive, the fourth drive, and the sixth drive, the remaining drive range disposed between the gear pairs on both sides is only the second drive and therefore there arises no problem about an arrangement order with respect to this remaining drive range. On the other hand, in the case of disposing a plurality of remaining drive ranges between the gear pairs on both sides, it is a matter of course that gear pairs of these drive ranges should be arranged such that the gear pair of the higher-speed drive range is disposed closer to the engine in view of the foregoing requirement that the countershaft 15 be reduced in diameter from its intermediate portion toward its front end.

[0029]

In the illustrated example, the gear 33 on the countershaft 15 forming the second drive gear pair G2 has a larger diameter than the gear 35 on the countershaft 15 forming the fourth drive gear pair G4 so that the outer diameter of the countershaft 15 at its portion where the gear 33 is disposed tends to be greater than the outer diameter thereof at its portion where the gear 35 is disposed. However, such a tendency can be easily solved by a measure of interposing an annular spacer 36 (see Fig. 2) between the gear 33 and the countershaft 15, or the like. Therefore, it does not follow that the foregoing requirement for reducing the diameter of the countershaft 15 from its intermediate portion toward its front end

cannot be satisfied due to the foregoing arrangement of the gear pairs.

[0030]

The countershaft 15 is further provided with a sixth drive-dedicated synchromesh mechanism 37 disposed between the sixth drive output gear 31 and the second drive output gear 33.

When a coupling sleeve 37a thereof is moved leftward from an illustrated neutral position to be meshed with a clutch gear 37b, the sixth drive output gear 31 is drivingly connected to the countershaft 15 so that the sixth drive becomes selectable as will be described later.

Further, the countershaft 15 is provided with a second drive-fourth drive synchromesh mechanism 38 disposed between the second drive output gear 33 and the fourth drive output gear 35.

When a coupling sleeve 38a thereof is moved leftward from an illustrated neutral position to be meshed with a clutch gear 38b, the second drive output gear 33 is drivingly connected to the countershaft 15 so that the second drive becomes selectable as will be described later, and

when the coupling sleeve 38a is moved rightward from the illustrated neutral position to be meshed with a clutch gear 38c, the fourth drive output gear 35 is driving connected to the countershaft 15 so that the fourth drive becomes selectable as will be described later.

[0031]

A description will be given next about the operation of the twin-clutch manual transmission according to the foregoing embodiment.

In a neutral (N) range or a parking (P) range where no power transmission is desired, the clutches C1 and C2 are both engaged, but the coupling sleeves 28a, 29a, 37a, and 38a of the synchromesh mechanisms 28, 29, 37, and 38 are all set in the illustrated neutral positions, thereby causing the twin-clutch manual transmission not to carry out the power transmission.

In a D range where forward power transmission is desired or an R range where reverse power transmission is desired, each of the forward drive ranges or the reverse drive range can be selected by controlling the coupling sleeves 28a, 29a, 37a, and 38a of the synchromesh mechanisms 28, 29, 37, and 38 and the clutches C1 and C2 in the following manner by the use of the hydraulic oil from the oil pump 4 as a medium.

[0032]

When the first drive is desired in the D range, the clutch C1 in the engaged state is released and the coupling sleeve 28a of the synchromesh mechanism 28 is moved leftward to drivingly connect the gear 22 to the countershaft 15, and thereafter, the clutch C1 is engaged.

Through this operation, engine rotation from the clutch C1 is output from the output shaft 11 in the axial

direction via the first input shaft 5, the first drive gear pair G1, the countershaft 15, and the output gear pair 19, 20 so that the power transmission in the first drive can be achieved.

When the selection of the first drive is for starting, an engagement control of the clutch C1 therefor is, of course, carried out.

[0033]

When upshifting from the first drive to the second drive, the clutch C2 in the engaged state is released and the coupling sleeve 38a of the synchromesh mechanism 38 is moved leftward to drivingly connect the gear 33 to the countershaft 15, and thereafter, the clutch C1 is released while the clutch C2 is engaged (switching of the clutches), thereby effecting the upshift from the first drive to the second drive.

After the completion of the upshift, the coupling sleeve 28a of the synchromesh mechanism 28 is returned to the neutral position to thereby separate the gear 22 from the countershaft 15, and thereafter, the clutch C1 is placed in the engaged state.

Through this operation, engine rotation from the clutch C2 is output from the output shaft 11 in the axial direction via the second input shaft 6, the second drive gear pair G2, the countershaft 15, and the output gear pair 19, 20 so that the power transmission in the second drive can be achieved.

[0034]

When upshifting from the second drive to the third drive, the clutch C1 in the engaged state is released and the coupling sleeve 29a of the synchromesh mechanism 29 is moved leftward to drivingly connect the gear 26 to the first input shaft 5, and thereafter, the clutch C2 is released while the clutch C1 is engaged (switching of the clutches), thereby effecting the upshift from the second drive to the third drive.

After the completion of the upshift, the coupling sleeve 38a of the synchromesh mechanism 38 is returned to the neutral position to thereby separate the gear 33 from the countershaft 15, and thereafter, the clutch C2 is placed in the engaged state.

Through this operation, engine rotation from the clutch C1 is output from the output shaft 11 in the axial direction via the first input shaft 5, the third drive gear pair G3, the countershaft 15, and the output gear pair 19, 20 so that the power transmission in the third drive can be achieved.

[0035]

When upshifting from the third drive to the fourth drive, the clutch C2 in the engaged state is released and the coupling sleeve 38a of the synchromesh mechanism 38 is moved rightward to drivingly connect the gear 35 to the countershaft 15, and thereafter, the clutch C1 is released while the clutch C2 is engaged (switching of the clutches),

thereby effecting the upshift from the third drive to the fourth drive.

After the completion of the upshift, the coupling sleeve 29a of the synchromesh mechanism 29 is returned to the neutral position to thereby separate the gear 26 from the first input shaft 5, and thereafter, the clutch C1 is placed in the engaged state.

Through this operation, engine rotation from the clutch C2 is output from the output shaft 11 in the axial direction via the second input shaft 6, the fourth drive gear pair G4, the countershaft 15, and the output gear pair 19, 20 so that the power transmission in the fourth drive can be achieved.

[0036]

When upshifting from the fourth drive to the fifth drive, the clutch C1 in the engaged state is released and the coupling sleeve 29a of the synchromesh mechanism 29 is moved rightward to directly connect the first input shaft 5 to the output shaft 11, and thereafter, the clutch C2 is released while the clutch C1 is engaged (switching of the clutches), thereby effecting the upshift from the fourth drive to the fifth drive.

After the completion of the upshift, the coupling sleeve 38a of the synchromesh mechanism 38 is returned to the neutral position to thereby separate the gear 35 from the countershaft 15, and thereafter, the clutch C2 is placed in the engaged state.

Through this operation, engine rotation from the clutch C1 is output from the output shaft 11 in the axial direction via the first input shaft 5 and the coupling sleeve 29a so that the power transmission in the fifth drive (gear ratio 1:1) can be achieved.

[0037]

When upshifting from the fifth drive to the sixth drive, the clutch C2 in the engaged state is released and the coupling sleeve 37a of the synchromesh mechanism 37 is moved leftward to drivingly connect the gear 31 to the countershaft 15, and thereafter, the clutch C1 is released while the clutch C2 is engaged (switching of the clutches), thereby effecting the upshift from the fifth drive to the sixth drive.

After the completion of the upshift, the coupling sleeve 29a of the synchromesh mechanism 29 is returned to the neutral position to thereby release the direct connection between the first input shaft 5 and the output shaft 11, and thereafter, the clutch C1 is placed in the engaged state.

Through this operation, engine rotation from the clutch C2 is output from the output shaft 11 in the axial direction via the second input shaft 6, the sixth drive gear pair G6, the countershaft 15, and the output gear pair 19, 20 so that the power transmission in the sixth drive can be achieved.

[0038]

When downshifting from the sixth drive toward the first drive in order, a predetermined downshift can be effected by performing a control reverse to the foregoing upshift.

[0039]

In the R range where reverse power transmission is desired, the clutch C1 engaged in the N range is released and the coupling sleeve 28a of the synchromesh mechanism 28 is moved rightward to drivingly connect the gear 24 to the countershaft 15, and thereafter, the clutch C1 is engaged.

Through this operation, engine rotation from the clutch C1 is output from the output shaft 11 in the axial direction via the first input shaft 5, the reverse gear pair GR, the countershaft 15, and the output gear pair 19, 20. In this event, since the rotation direction is reversed by the reverse gear pair GR, the power transmission in the reverse drive range can be achieved.

At the time of starting in the reverse drive range, an engagement control of the clutch C1 therefor is, of course, carried out.

[0040]

According to the twin-clutch manual transmission of this embodiment having the foregoing structure, in the arrangement of the gear pairs G2, G4, and G6 of the even-number drive range group (second drive, fourth drive, sixth drive) provided between the second input shaft 6 and the countershaft 15,

since the gear pair G4 of the lowest-speed drive range (fourth drive) in the drive ranges (fourth drive, sixth drive) relating to the gears 30 and 34 each having an outer diameter capable of providing the bearing reception space for the needle bearing 8 interposed between the first input shaft 5 and the second input shaft 6, among the gears 30, 32, and 34 on the second input shaft 6, is disposed on the side farthest from the engine,

it is possible to ensure the needle bearing (8) reception space between the rear end of the second input shaft 6 and the first input shaft 5 without forming the needle bearing (8) reception annular groove on the first input shaft 5 so that it becomes possible to receive the needle bearing 8 between the rear end of the second input shaft 6 and the first input shaft 5 without lowering the strength of the first input shaft 5.

Therefore, according to this embodiment, the bearing span of the needle bearings 7 and 8 provided between both input shafts 5 and 6 can be set large to thereby achieve high bearing rigidity between both input shafts 5 and 6.

[0041]

Further, according to this embodiment, in the arrangement of the gear pairs G2, G4, and G6 of the even-number drive range group (second drive, fourth drive, sixth drive) provided between the second input shaft 6 and the countershaft 15, the gear pair G6 for the highest-speed drive range (sixth drive) in the drive ranges (second drive,

sixth drive) other than the foregoing fourth drive is disposed on the side closest to the engine,

so that the front end of the countershaft 15 near the engine can be made small in diameter since the gear 31 on the countershaft 15 forming the gear pair G6 has a small diameter because of it being for the highest-speed drive range,

and therefore, it is possible to satisfy the foregoing requirement in terms of the assembly and also the strength that the diameter of the countershaft 15 be configured to decrease from its intermediate portion toward its front end.

[0042]

In the case where the number of the drive ranges of the drive range group provided between the second input shaft 6 and the countershaft 15 is more than the illustrated case of three so that a plurality of drive ranges are provided between the gear pairs disposed on the side farthest from the engine and the side closest to the engine, the foregoing requirement for reducing the diameter of the countershaft 15 from its intermediate portion toward its front end can be satisfied by arranging gear pairs of these plurality of drive ranges such that the gear pair of the higher-speed drive range is disposed closer to the engine.

[0043]

Further, it has been confirmed that in the case where

the drive range group provided between the second input shaft 6 and the countershaft 15 is the even-number drive range group like in the illustrated example, even when the number of the even-number drive ranges is not only three like in the illustrated example but is more than three, the fourth drive, like in the illustrated example, satisfies the foregoing requirement and the disposition of the fourth drive gear pair G4 on the side farthest from the engine is practical in terms of a relationship with practically suitable gear ratios.

[0044]

Further, in this embodiment, since the synchromesh mechanisms 38 and 37 for causing the gear pairs G2, G4, and G6 of the drive range group (second drive, fourth drive, sixth drive) provided between the second input shaft 6 and the countershaft 15 to be properly transmittable, respectively, are all disposed on the side of the countershaft 15,

it is not necessary to provide the synchromesh mechanisms 38 and 37 on the second input shaft 6 that tends to be thin in thickness because of it being hollow and due to restriction in radial space so that a reduction in rigidity of the second input shaft 6 can be avoided, which is highly advantageous.

[0045]

Moreover, in this embodiment, since the mesh mechanism 37 dedicated for causing the gear pair G6

disposed on the side closest to the engine to be properly transmittable is disposed between the gear pair G6 disposed on the side closest to the engine and the gear pair G2 disposed adjacent thereto among the gear pairs G2, G4, and G6 of the drive range group (second drive, fourth drive, sixth drive) provided between the second input shaft 6 and the countershaft 15,

a synchromesh structural member including a clutch gear such as the clutch gear 37b does not exist between the mesh mechanism 37 and the gear pair G2 (gear 33) so that, correspondingly thereto, the gear pair G2 (gear 33) can be disposed closer to the shaft support portion (roller bearing) 16 for the countershaft 15 with respect to the transmission case front wall 1a, and therefore, support rigidity for the gear pair G2 (gear 33) that transmits a large torque because of its large reduction ratio can be set large enough to bear such a large torque.

[0046]

In the case where, like in the illustrated example, the gear pairs G2, G4, and G6 of the drive range group (second drive, fourth drive, sixth drive) provided between the second input shaft 6 and the countershaft 15 are disposed as described above, the mesh mechanisms 38 and 37 for causing the gear pairs G2, G4, and G6 to be properly transmittable are disposed on the side of the countershaft 15, and, as the mesh mechanisms disposed on the side of the countershaft 15, the mesh mechanism 38 for both the second

drive and the fourth drive is disposed between the second drive gear pair G2 and the fourth drive gear pair G4 and the mesh mechanism 37 dedicated for the sixth drive is disposed between the second drive gear pair G2 and the sixth drive gear pair G6, the foregoing various operations and effects can all be achieved, which is highly advantageous as the FR-vehicle six-speed twin-clutch manual transmission.

[Brief Description of the Drawings]

[0047]

[Fig. 1] A diagram of essentials of a twin-clutch manual transmission according to one embodiment of the present invention.

[Fig. 2] A longitudinal sectional side view showing a substantial structure of the twin-clutch manual transmission.

[Description of Symbols]

[0048]

1	transmission case
1a	transmission case front wall
1b	transmission case intermediate wall
1c	transmission case rear wall
2	engine crankshaft
C1	odd-number drive range clutch
C2	even-number drive range clutch
3	torsional damper
4	oil pump

5	first input shaft
5a	first input shaft rear-end portion
6	second input shaft
7	front-side needle bearing
8	rear-side needle bearing
11	output shaft
15	countershaft
19	counter gear
20	output gear
G1	first drive gear pair
G2	second drive gear pair
G3	third drive gear pair
G4	fourth drive gear pair
G6	sixth drive gear pair
GR	reverse gear pair
28	first drive-reverse synchromesh mechanism
29	third drive-fifth drive synchromesh mechanism
37	sixth drive synchromesh mechanism
38	second drive-fourth drive synchromesh mechanism

[Name of Document] ABSTRACT

[Abstract]

[Object] To propose an arrangement of gear pairs wherein, of bearings between first and second input shafts in the form of a dual shaft, the rear bearing can be disposed at a rear end of the outer-side second input shaft to thereby increase a bearing span, and a countershaft can be formed into a shape tapering toward its front end.

[Solving Means] In arranging gear pairs G2, G4, and G6 of a drive range group (second drive, fourth drive, sixth drive) provided between a second input shaft 6 enclosing a first input shaft 5 and a countershaft 15, the gear pair G4 of the lowest-speed drive range (fourth drive) in the drive ranges (fourth drive, sixth drive) relating to gears 30 and 34 each having an outer diameter capable of providing a bearing reception space for a needle bearing 8 interposed between the first input shaft 5 and the second input shaft 6, among the gears 30, 32, and 34 on the second input shaft 6, is disposed on the side farthest from an engine, the gear pair G6 for the highest-speed drive range (sixth drive) in the drive ranges (second drive, sixth drive) other than the fourth drive is disposed on the side closest to the engine, and the remaining second drive gear pair G2 is disposed between both ends of the second input shaft 6. It is possible to ensure the needle bearing (8) reception space between a rear end of the second input shaft 6 and the first input shaft 5 so that the bearing span between

the needle bearings 7 and 8 can be set large, and further, the countershaft 15 can be formed into a shape tapering toward its front end.

[Selected Figure] Fig. 2